Fluid-Meter Nozzles[1]

Ronald B. Smith.[2] The problem of establishing reliable nozzle coefficients and a standard flow-measuring technique for acceptance-test work is of particular concern to the Power Test Codes Committee at the present time. In the writer's opinion the shape that is chosen matters but little providing only that the nozzle can be easily reproduced and accurately installed. The important point is to choose a standard which has been so thoroughly verified that its characteristics under all probable test installations are accurately established.

One of the nozzles under consideration as a standard is the V.D.I. profile, some of the characteristics of which Mr. Buckland compares with the G.E. nozzle. The V.D.I. nozzle is the outgrowth of the nozzle used for the past 30 years for flow-measurement work by the I. G. Farbenindustrie. Within recent times it has been adopted as standard by the International Standards Association and now, as a result, is generally known as the I.S.A. nozzle. Since 1928, largely at the request of the V.D.I., several thousand laboratory calibrations of the nozzle have been made, with the result that its flow coefficients, for pressure drops down to the acoustic and for area ratios ranging from zero to 0.6, have been established under a wide variety of conditions with an accuracy generally within plus or minus 0.5 per cent. The flow coefficient of the I.S.A. nozzle is constant over a greater range than is usually the case with a full flowing nozzle. For instance, in the author's Fig. 14 it is evident that the I.S.A. nozzle can be used to a 50 per cent lower range than the G.E. nozzle before one must resort to cut-and-try methods in the calculations.

By attempting to compensate for the different locations of the pressure taps the author concludes that the coefficient of the G. E. nozzle is $1^1/_4$ per cent higher than the I.S.A. This result is based on the assumption that a pressure measurement in the throat of a nozzle and a pressure measurement in the pipe two nozzle diameters downstream are identical, and are equivalent to the atmospheric pressure with a freely discharging jet. This opinion appears untenable from analysis of the very tests that the author quotes to support it, namely, the work of Stach. Except for the smallest nozzle, Stach's coefficients show less than 0.3 per cent difference between measurements of the I.S.A. nozzle when discharging freely and when operating in a pipe with the usual corner taps.[3] This slight difference can be explained by the fact that Stach used pressure-chamber openings smaller than standard. Thus, the result of Stach's work is to indicate that the pressure measurement in the downstream corner is equivalent to the pressure for discharge into an infinite chamber.

If we compare the pipe and throat-tap measurements on a Moss-Johnson nozzle as reported by Sprenkle,[4] we must conclude that a throat tap reads the pressure about $1^1/_2$ per cent high.

Downstream from the nozzle and along the pipe wall there is a pressure fall. It is interesting to note that this distribution, as in Fig. 1 of this discussion, on a Moss-Johnson nozzle which is practically the same as the G.E. nozzle, is similar to the results reported by Witte on the I.S.A. nozzle. For instance, the minimum pressure is $1^1/_2$ per cent of the differential pressure and it occurs (for $m = 0.25$) about $^3/_4$ diam downstream from the mouth. Because the same quantitative phenomenon is observed with an orifice it seems probable that the distribution is produced by viscous effects at the boundary of the jet.

In combination with Sprenkle's results, the pressure-distribution curve leads one to suspect that throat pressures are 1 per cent higher than corner pressures, and that, as a result, if the nozzles are compared on this basis, there will be practically no difference in their coefficients. That there is no actual difference between the two nozzle coefficients when the pressures are measured in the

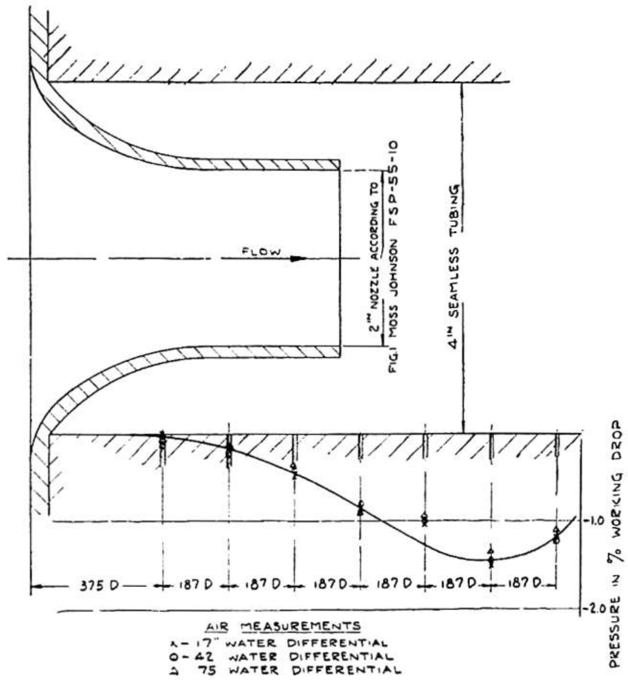

Fig. 1 Pressure Distribution in a Moss-Johnson Flow Nozzle

same manner has been proved by Witte.[3] For instance, using corner taps, which includes some impact pressure and therefore accounts for a low coefficient, he measures with an area ratio 0.25:

For the I.S.A. nozzle	=	0.9765
For the Moss-Johnson nozzle	=	0.974
For the same Moss-Johnson nozzle with throat taps.	=	0.991

Thus the relative magnitudes in the author's Fig. 14 are misleading.

Mr. Buckland suggests that the rapid fall of the coefficients of the I.S.A. nozzle below the operating region is the result of contact loss in the nozzle. That there is a temporary contact loss in an I.S.A. nozzle where the approach radius is tangent to the throat is known. Not so well known is the fact that nozzles of the G.E. shape also show contact loss at this point. I have observed the phenomenon many times on a Moss-Johnson nozzle by coating the inner surface with lampblack and kerosene and studying the streak lines that are produced by the air. However, in neither nozzle is it of serious importance nor would it be termed a vena contracta, since contact is reestablished within $^1/_4$ in. downstream. The peculiar slope of Mr. Buckland's coefficients in Fig. 8 of his paper between Reynolds' numbers of 10^4 and 4×10^5 may be the result of this phenomenon.

R. E. Sprenkle.[5] Mr. Buckland's paper will be of material assistance in familiarizing engineers with the fact that the Ameri-

[1] Published as paper FSP-56-14 by B. O. Buckland, in the November, 1934, issue of the A.S.M.E. Transactions.

[2] Turbine Engineering Department, Westinghouse Electric and Manufacturing Company, South Philadelphia, Pa. Jun. A.S.M.E.

[3] "Neuere Mengenstrommessung zur Normung von Dusen und Blenden," by R. Witte, *Forschung auf dem Gebiete des Ingenieurwesens*, September-October, 1934.

[4] "A System for the Measurement of Steam With Flow Nozzles for Turbine Performance Tests," by S. A. Moss and W. W. Johnson, Trans. A.S.M.E., vol. 55, 1933, paper FSP-55-10 p. 145.

[5] Bailey Meter Company, Cleveland, Ohio. Assoc-Mem. A.S.M.E.

can style of flow nozzle is a real precision instrument and that its accuracy and reliability well merit its use as a standard of measurement.

Our experience in building and using several thousand flow nozzles of all sizes and for all kinds of flow-metering service, has shown that the use of pipe-line connections at both the nozzle inlet and outlet, as shown in Fig. 13 of the paper, is the simplest and most dependable method of measuring the pressure differential across the nozzle. This experience covers a span of nearly twenty years during which many weighed-water or other tests have proved the adequacy of the commercial nozzle of this design as a means for measuring water, steam, air, gas, and other flow rates.

Pipe taps into the wall back of the nozzle throat instead of into the throat itself, possess some real advantages. First, this location is in a protected zone out of the path or contact with the stream lines of the flowing fluid, and thus not susceptible to errors in static-pressure measurement due to small localized eddies, whirls, or other disturbances such as may, and often do, exist along the throat surface. Moreover, being in a zone where velocities are comparatively low, there is even less chance of this pressure measurement being in error.

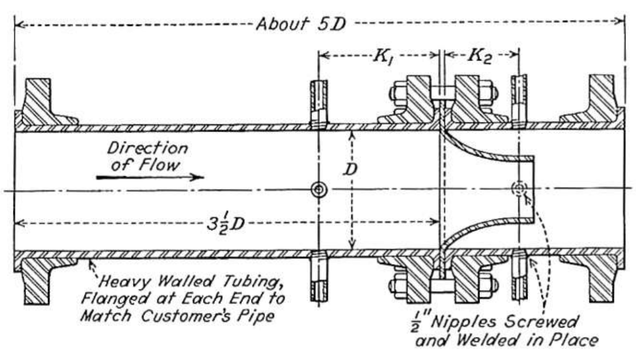

Fig. 2 A Typical Nozzle Pipe Section With Pressure Connections and Nozzle Properly Located

From the standpoint of physical application, the nozzle with the outlet connection made back of, instead of into, the throat, allows the use of a much thinner flange, with a consequent reduction in pipe spread to provide for its insertion between existing flanges. The nozzle proper is easier to build because of the omission of the piezometer chamber and internal connection passages which are required to provide for throat taps.

One of the outstanding advantages of the nozzle, shown in the author's Fig. 13, is that extreme care does not need to be taken in drilling the outlet connection into the pipe wall. True, this must be done in the field, as Mr. Buckland states, but due to its protected location, it is much less difficult to make than the inlet pipe connection which is used by both nozzle types.

The throat tap connection is admittedly difficult to make unless all possible precautions are taken. This point is brought out not only by Mr. Buckland himself, but also in the discussion of the Moss-Johnson paper in 1932 by the present writer, in which comparative tests made with both throat and pipe taps in a special nozzle in the Bailey Meter Company laboratory were described in detail. In that discussion, we demonstrated the difficulty, in fact, almost impossibility, of getting the separate throat-tap pressure readings to check each other, as compared with the ease of obtaining a very satisfactory agreement between the different outlet pipe tap readings. The elimination of this job of making satisfactory throat connections more than compensates for the labor of providing for this connection in the field.

While the pipe tap back of the throat cannot be calibrated as an integral part of the nozzle itself, neither can the inlet-pipe connection which is used with both types of nozzles. And of the two, the inlet connection is the more susceptible to changes in the flow state, being immediately adjacent to the path of the stream lines. As such, it is the most important connection to be included in any integral nozzle-assembly calibration. The truly correct and proper method is to calibrate the nozzle with the section of pipe in which it is to be used, and thus both pressure connections are included in the assembly and all possible installation vagaries eliminated. A typical nozzle pipe section with pressure connections and nozzle properly located, is shown in Fig. 2 of this discussion.

The first reason given for the use of the throat instead of the pipe-line connections, was that the lack of geometrical similarity of the external shape of the nozzle used might produce erroneous results were pipe taps used. We would point out that even along the internal surfaces over which the fluid passed, complete geometrical similarity did not exist. True, the test nozzles were in themselves, geometrically similar in form, but when installed in the pipe lines, the assemblies with the pipe were not geometrically similar by widely varying amounts. To attain complete similarity, the curvature must begin at the same relative point with reference to the inside pipe wall on each nozzle. In all but one of the G.E. nozzles, the distance of the junction of the curvature with the straight flange section as measured from the inside of the pipe wall, varied from 4 per cent to 27 per cent of pipe diameter, and in this one case, this point was actually up in the holding flange by an amount equal approximately to 5 per cent of the pipe diameter. Since the flow must pass over these surfaces, this lack of similarity is likely to produce a larger spread between coefficients of different nozzles of various sizes than would have resulted from pipe-tap measurements made in a region where this lack of similarity was relatively unimportant.

That there can be no *complete* geometrical similarity between nozzles of different diameter ratios, is quite apparent but nevertheless not always fully understood. In fact, only when nozzles of the same diameter ratio are used in different sizes of pipes can such similarity be obtained, and even then the relative pipe roughness may not be quite the same. Since various diameter-ratio sizes must be used for practical metering, it is useless to expect complete agreement of calibration data on a similarity basis.

An improvement can be made, in the attaining of better similarity between different diameter-ratio sizes, by always placing the beginning of the curvature at the surface of the internal pipe wall and then so shape the curvature to the one-quarter ellipse by making the minor axis equal to $(D-d)/2$ instead of $5/8\ d$. With increasing diameter ratios, this ellipse becomes flatter but there are no humps or irregular surfaces over which the fluid must flow and thus no marked or sudden deviations from the natural flow path.

A comparison of calibration data from nozzles of the shape just described, using pipe taps, with the General Electric nozzles would be of interest. In Fig. 11, Mr. Buckland shows such a comparison between the data from a 12-in. × 7.554-in. Bailey Meter Company nozzle, and the average G.E. curve. The agreement of one with the other, is about as perfect as can be expected. However, the range of Reynolds' number used with the 12-in. nozzle was rather small; so to extend the curve to lower limits, a 3-in. pipe size, 60 per cent diameter-ratio nozzle was recently calibrated in our laboratory in Cleveland.

The calibration data obtained from a Bailey Meter Company 3.06-in. × 1.836-in. nozzle is shown in Fig. 3 of this discussion, as well as that from the 12-in. × 7.554-in. nozzle. In passing,

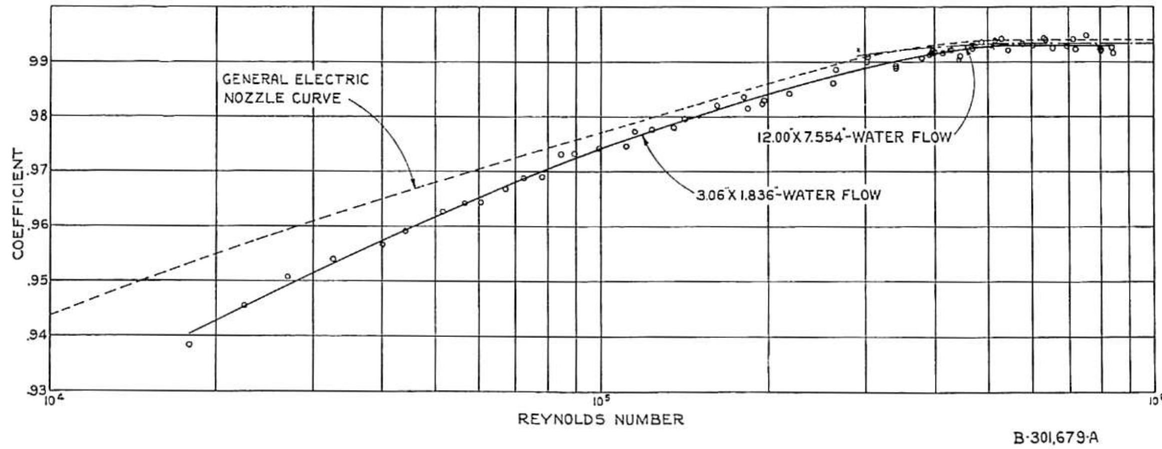

Fig. 3 Calibration of Two Bailey Meter Company Flow Nozzles Compared With Average Curve of General Electric Nozzles

it should be noted that the 3-in. nozzle was made of highly polished brass and calibrated in smooth brass tubing, using water flow while the 12-in. nozzle was made of steel and calibrated in a commercial steel tubing, also using water as the flowing fluid. As will be noted, these two nozzles, checked each other almost perfectly through a range of Reynolds' number from 500,000 to the highest point calibrated and deviated from each other at the most about $^1/_4$ per cent at a Reynolds number of 300,000 or the lowest point tested on the 12-in. nozzle.

In Fig. 3 of this discussion, we have also shown by the broken line, the average calibration curve from Mr. Buckland's Fig. 8, as a comparison with calibrations of two Bailey Meter Company nozzles. Despite the higher diameter ratio, and the fact that both the Bailey nozzles shown used pipe-line pressure connections back of the nozzle throat instead of into the throat, and further, that the internal contour of the nozzle shape was not precisely the same, the General Electric type and the Bailey type checked each other from 0.1 per cent to 0.3 per cent over a working range of Reynolds' number of from 100,000 to 1,500,000, or the highest tested point. Whether or not the increased deviation at lower values of Reynolds' number is due to the difference between the location of the outlet pressure connections, to the small difference in the shape, or to experimental errors, is a question we cannot adequately answer at this time.

It is sufficient to add that through the useful range of Reynolds' number, or from 100,000 up, and with nozzles of diameter ratios not materially exceeding 60 per cent, either of the two types of American nozzles can be used for the purposes outlined in this paper with an accuracy that is certainly well within plus or minus 1 per cent, and with actual calibration within plus or minus $^1/_2$ per cent, provided proper precautions are taken both as to obtaining undisturbed flow through the nozzle, and in the design, construction, and installation of the nozzle assembly itself.

R. J. S. Pigott.[6] In examining Mr. Buckland's paper, one notes a tendency to call various nozzles, or orifices, geometrically similar, when as a matter of fact, they are not. It is not enough to use nozzles that are similar in contour because, for rigorous comparisons, it is necessary also to have similarity in pressure taps, polish of nozzle, upstream pipe, and orifice ratio. This complete condition has practically never been observed in tests as yet, and until it is, we shall not be able to get the full value of Reynolds' criterion comparisons. Scatter of points is much wider than can be assigned correctly to experimental errors, or to any departures from the "single line" theory; and the whole situation for studying the proper relations is somewhat confused.

The long series of experiments on orifices conducted by the joint A.G.A.-A.S.M.E. meter committee shows that the upstream roughness, orifice ratio, and tap location have very noticeable effects upon coefficient, quite in line with theory. While some advance the thought that pipe roughness has no effect on a nozzle, theory clearly indicates there ought to be some effect. If this thought were correct, neither the sharp-edged orifice nor the venturi should show roughness effect; but we know that they do show such differences.

Mr. Buckland has recognized this point in his Fig. 9 wherein the curves better approach full similarity, by eliminating orifice-ratio effects; pipe and nozzle relative roughness remaining the same.

The writer has been working for some time on a method of predicting coefficients, and finds that there is a definite relation between the ratio of "surface area" washed by the fluid between taps, to the area of throat, and the coefficient at any Reynolds number. The relative loss, or $(1 - C)$, is directly related to the pipe flow friction. The writer has for some time used the coordinates $(1 - C)$ vs. $dv\rho/\mu$ on double-log paper. Mr. Ed Smith has also used the same type of coordinates. It gives some very very valuable analytical indications which the semi-log graph of C vs. $dv\rho/\mu$ is incapable of showing.

One other point in nozzle testing has not been given sufficient attention. On a curve showing throat Reynolds' number, we would expect complete viscous flow below $R = 1200$. But above that point, the nozzle is in mixed flow until the upstream section also is fully turbulent. With an orifice ratio of 0.50, the minimum value of throat R for complete turbulence is 2500, and higher for smaller ratios. In addition, there is apparently a stronger tendency for a convergent nozzle to stay in the viscous region at higher values than in parallel sided pipe. As a consequence, many nozzles tested by Mr. Buckland and others cannot be safely considered in fully turbulent flow until values of possibly R = 30,000 to 40,000 have been passed. There is, therefore, a considerable range in which the flow is somewhat unstable, and the scatter of test points will usually be a little wider.

With regard to a supposed critical Reynolds' number at which the coefficient becomes constant, the writer is inclined to doubt any such value of R as 10^5. In pipe flow, such flattening does not take place until $R = 2 \times 10^6$ to 4×10^6. What appears to be a flat coefficient is merely due to rate of change much smaller than the test accuracy can show. A logarithmic graph

[6] Staff Engineer, Charge of Engineering, Gulf Research & Development Corp., Pittsburgh, Pa. Mem. A.S.M.E.

of (1 — C) shows this condition very plainly. A sloping line at 11 deg or 12 deg, corresponding to the smooth-pipe conditions, will fit this cloud of points quite as well as a horizontal line. In this region, a precision of plus or minus 0.5 per cent in the tests means a variation of 40 to 80 per cent of (1 — C). It is, of course, futile at present to attempt to prove this point, until still better test accuracy can be attained.

The A.S.M.E. Special Research Committee is undertaking an extensive program of investigation on this subject, with the original intention of comparing the proposed I.S.A. or Witte nozzle, with the type discussed in Mr. Buckland's paper. In order to determine more closely those factors not too clearly defined at present, such as roughness and diameter-ratio effects, the program will cover full-range tests on a preferred-number series of both sizes and ratios, with geometric similarity as fully developed as possible. Tests will be made in full with water, but duplicated so far as necessary with steam and air. Funds for this work are to be collected, as is usual in A.S.M.E. research undertakings, from interested industries.

W. S. Cooper.[7] The writer believes that Mr. Buckland should have given more data on the performance of these nozzles in actual field tests. High order of accuracy in measuring flow-rates is demanded on acceptance tests by builders and purchasers of turbine and boiler-room equipment, and direct measurement (by weighing) of condensate and feed-water flow rates has heretofore been considered the only reliable means. After all, the field of application of the nozzle will lie in the replacement of the more expensive direct-weighing method, and it is under such circumstances that a knowledge of the nozzle's performance characteristics is desired.

There is doubt in the writer's mind as to whether such accuracy as claimed by the author with laboratory tests could be obtained with the piping situation usually encountered in the average power plant. Furthermore, liquid flow in most power-plant piping is of a pulsating nature since the fluids are handled either by centrifugal or by reciprocating pumps. Pulsation was probably entirely absent or eliminated in the laboratory tests where the fluid was probably supplied by standpipes.

The writer had occasion recently to conduct field tests on one of the General Electric Company's nozzles described by the author. This nozzle was the one with proportions shown in the seventh line of Table 1 in Mr. Buckland's paper, namely, the 12.01-in. × 5.016-in. nozzle. The laboratory test results reported by the author for this nozzle are shown in his Fig. 8, the plotted points appearing as plus signs. The writer's tests were conducted in conjunction with two condenser acceptance tests where the main condensate was weighed with an accuracy within 0.1 per cent on carefully calibrated scales.

The nozzle was inserted in series with the 12-in. main condensate test header which delivered the condensate from the condenser under test to the weighing tanks. With respect to the piping, the nozzle was located in as favorable a situation as will be found in the field. The nozzle was inserted at a point corresponding to about 110 ft of approach piping (which would tend to minimize pulsation) and the nozzle itself was preceded by 14 ft of straight piping of uniform size. The downstream side of the nozzle consisted of 7$^1/_4$ ft before the first obstruction was reached.

The pressure differential across the nozzle was read from two mercurial single-column cistern-type manometers. Both manometers were connected to the same upstream static-pressure tap located in a horizontal plane 12 in. before the entrance edge of the nozzle. The low-pressure side of one manometer was connected through an internal port to the piezometer ring in the throat of the nozzle, while the low-pressure side of the other manometer was connected to a downstream static tap in the pipe at a transverse plane passing through the discharge end of the nozzle. This double arrangement was furnished to provide check readings of the nozzle differential. Each manometer was read by a separate observer. It was found that there was practically no difference between the two sets of readings.

The results of the writer's test are shown in Fig. 4 of this discussion. The horizontal line at C = 0.994 is that portion of the author's blanket curve from his Fig. 8 that applies to the range of Reynolds' number used by the writer. The plotted points indicate the spread of the nozzle coefficient, C, as determined under field conditions. The estimated maximum error on this field test is about 2.4 per cent and the estimated probable error is 0.5 per cent. It should be noted, however, that these field results

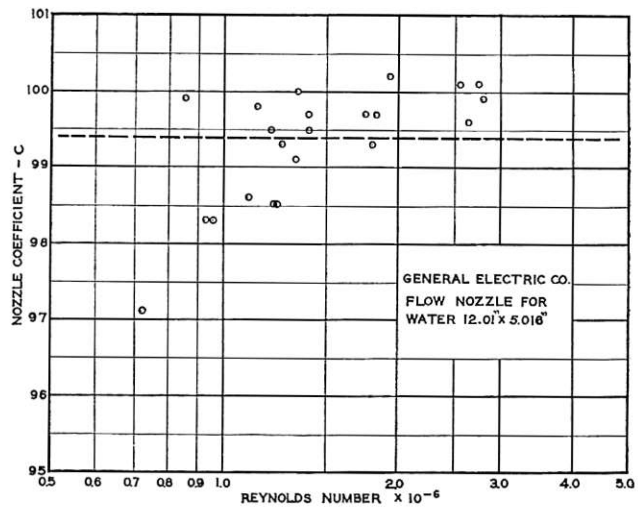

Fig. 4 Performance of General Electric Company Flow Nozzle During a Field Test

(Each point represents a 1$^1/_2$-hr run during which the main condensate from a steam condenser was also very accurately weighed after passing through the nozzle. The dashed line at C = 0.994 is that portion of the curve in Mr. Buckland's Fig. 8, corresponding to the above field data. The flow nozzle tested in this field test was the same as the 12.01-in. × 5.016-in. nozzle tested by the author.)

were obtained under unusually favorable circumstances. It is probably true that in the average run of cases where the nozzle could be used, results would not be so reliable as in this case.

Sanford A. Moss[8] and W. W. Johnson.[9] There are of course a good many ways in which flow may be measured with laboratory precision, and the paper by Mr. Buckland is a good example of one of them. It is to be noted that the work was carried out with great care and with a test set-up especially made for the flow measurement, and with all details arranged so that certainty of accuracy was the primary consideration. This puts the work in a wholly different territory from flow measurement made by the usual commercial flow meter which must be suitable for permanent, simple installation and maintenance in a commercial pipe line with a small pressure drop, and with an instrument which gives direct reading of flow. None of these considerations can be allowed to influence the precise flow measurement with certainty of accuracy, which is the author's purpose.

Of course, future research may show that some of the details used by the author might be altered to give as nearly as possible,

[7] Research Bureau, Brooklyn Edison Company, Brooklyn, N. Y Assoc-Mem. A.S.M.E.

[8] Research Engineer, General Electric Company, West Lynn, Mass. Mem. A.S.M.E.

[9] Mechanical Engineer, General Electric Company, West Lynn, Mass. Mem. A.S.M.E.

measurements corresponding to theoretical flow. For instance, it might be that the throat taps should have a longer parallel portion, such as is shown in the author's Fig. 4 or Fig. 12. However, the close agreement of the author's points shows that his throat-tap measurements must be very good. His Fig. 9 shows a very close agreement between flow coefficients for different values of m, the ratio of nozzle to pipe area. This seems to indicate that the theoretical allowance for velocity of approach takes full account of the effect of pipe diameter, with the possible exception of the $^1/_5$ per cent mentioned which is much less than the errors of observation. The spread of the curves in Fig. 11 of Mr. Buckland's paper is evidence in the discussion as to whether or not Reynolds' number is a proper criterion for abscissas for flow coefficients for different conditions, as was discussed in the Moss-Johnson paper referred to by the author. We thought that our tests with different temperatures and pressures of steam were brought together better by using differential pressure divided by absolute initial pressure as abscissas, and it has also been proposed to use head as abscissas. The droop of the Moss-Johnson curve, No. 1 in Fig. 11, may be due to some such considerations. It must, of course, be admitted that it may also be due to observational irregularities because of the very small flow and small differential pressures at the beginning of the curve. Some of the data given in the Moss-Johnson paper seem to indicate that there was a definite difference in the coefficient with throat taps and with pipe taps, such as for the author's Fig. 13 and that with pipe taps the discharge coefficients are lower. This being the case, the flow coefficients for Fig. 13 seem very high.

E. D. Dickinson.[10] Mr. Buckland's paper is a confirmation of the principles behind the growing opinion that precise measurements of fluids can be obtained by the use of properly proportioned nozzles. The proportions of the nozzle itself constitute but one factor contributing to the accuracy of the results. It is essential that certain precautions be taken. When these precautions are taken, tests can be reproduced with absolute fidelity and the results can be depended upon to be as accurate as laboratory tests.

I do not hold a brief for any particular method of measuring flow by nozzles. However, I have relied upon the flow nozzle for obtaining accurate measurements of steam flow for a period of years and the results have confirmed my contention that precise measurements of flow can be obtained with greater reliability and at less cost by the use of a nozzle similar to that described by Mr. Buckland than by any other recognized method. We have run a great many tests, both of research nature and on commercial machines, where precise results were obtained and valuable information secured that could not have been possible had we not had at our disposal a calibrated flow nozzle similar to Mr. B. O. Buckland's and as described by Dr. S. A. Moss and Mr. W. W. Johnson in their paper[4] presented at the A.S.M.E. Annual Meeting in December, 1932.

Author's Closure

Ronald B. Smith states that Stach's tests (on the coefficients of the V.D.I. Normdüse discharging into the atmosphere) show less than 0.3 per cent difference between measurements of the I.S.A. nozzle when discharging freely and when operating in a pipe with the usual corner taps. He summarizes the situation by saying that the result of Stach's work is to indicate that the pressure measurement in the downstream corner is equivalent to the pressure for discharge into an infinite chamber.

I cannot agree with Mr. Smith's interpretation of Stach's data. The data show clearly that the coefficient of the Normdüse is the same when discharging into the atmosphere as it is when installed in a pipe when the downstream pressure is measured at the point of *minimum pressure on the pipe wall.* In order to clear up this point I shall reproduce Stach's calibration results together with Witte's measurements of pressure difference between the point of minimum pressure on the pipe wall and the downstream corner tap. Table 1 of this discussion shows flow coefficients and pressure differences taken from the papers by Witte and Stach.

TABLE 1 FLOW COEFFICIENTS AND PRESSURE DIFFERENCES ON THE V.D.I. NOZZLE AS GIVEN BY WITTE AND STACH

m	Press. diff. Δ (Witte)	α	α_a (Stach)	$\alpha_m = \alpha - \Delta/2$
0.10	0.010	0.989	0.984	0.984
0.20	0.016	0.999	0.993	0.991
0.30	0.020	1.016	1.010	1.006
0.40	0.023	1.045	1.036	1.034
0.50	0.026	1.096	1.078	1.083

m is the ratio of nozzle area to pipe area.
Δ is the difference between the downstream corner-tap pressure and the minimum pressure on the pipe wall expressed as a fraction of the difference between the up- and downstream corner-tap pressures.
α is the flow coefficient of the Normdüse in a pipe, using corner taps.
α_a is the flow coefficient when discharging into atmosphere, using upstream corner tap and the atmosphere.
α_m is the flow coefficient using the upstream corner tap and the minimum pressure on the pipe wall. It is obtained by subtracting $\Delta/2$ from α, since the fraction Δ is nearly twice as large as the difference produced in the coefficient by using the minimum pressure on the pipe wall instead of the corner-tap pressure.

A comparison of α_a and α_m shows them to be about equal, much more closely so than are α and α_a. I, therefore, conclude that the coefficient of the Normdüse is the same when discharging into the atmosphere as it is when installed in a pipe with the downstream pressure measured at the point of minimum pressure on the pipe wall.

Mr. Smith states that the relative magnitudes in Fig. 14 of the paper are misleading. As defined in the paper this figure is a comparison of the coefficient curve of the G. E. nozzle with the coefficient curve of the V.D.I. Normdüse. The V.D.I. nozzle coefficients have been corrected to what they would be if the upstream pressure had been measured one pipe diameter upstream from the nozzle face and the downstream pressure at the point of minimum pressure on the pipe wall. As Mr. Smith points out, Witte[11] compared the two nozzles by calibrating them both with corner pressure taps. Witte finds that under these conditions the coefficient of the G. E. nozzle is $^1/_5$ per cent lower than that of the V.D.I. nozzles. It is true, that in the light of these recent tests by Witte, Fig. 14 of the paper shows too large a difference between the two coefficients in the range where the coefficients are independent of Reynolds' number. It is also true that in accordance with the Bureau of Standards tests the magnitude of this difference shown in Fig. 14 is correct. However, whatever the correct relation between the coefficients may be in the range where they are independent of Reynolds' number, the same variation of the coefficients with Reynolds' number is given by Witte as is shown in Fig. 14. The coefficient of the V.D.I. nozzle rises much more abruptly with increasing Reynolds' number than that of the G.E. nozzles between Reynolds' numbers of 10^4 and 10^5.

It should be of interest to note in this connection that the usual conditions met in testing a 10,000-kw turbine will require the use of a flow nozzle about $1^1/_4$ in. in diameter in a 4 in. pipe, and the operation of the nozzle in a range of Reynolds' numbers from 10^4 to 10^5. This is very close to $m = 0.09$ and right in the range of the rapid rise of coefficient of the Normdüse. With Reynolds' numbers higher than this range, the Normdüse is as useful for flow measurements as any other carefully calibrated device but

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[11] "Neuere Mengenstrommessungen zur Normung von Düsen und Blenden," by R. Witte, *Forschung auf dem Gebiete des Ingenieurwesens*, September-October, 1934.

at this point of rapid rise in coefficient I would rather use a different nozzle.

I want to thank Mr. Smith for his data on the pressure distribution along the pipe wall on the Moss-Johnson nozzle.

I would like to thank Mr. Sprenkle for his data on the 3 × 1.8-in. nozzle. It would be interesting to have the dimensions showing the location of Mr. Sprenkle's pressure taps so that his data could be more readily compared with other available data.

I want to thank Mr. Pigott for his suggestion regarding a method of plotting flow coefficients.

In answer to Mr. Cooper's question regarding the use of the nozzles in the field, I have made seven turbine-performance tests in which the flow was measured only by means of flow nozzles. In these tests none of the Btu-rate or water-rate points scattered from an average curve more than ± 1/2 per cent. Each of these tests consists of approximately 15 or more points. These results obtained by the use of flow nozzles are much more satisfactory than the example shown by Mr. Cooper.

In the case he cites (The performance tests of the turbine units Nos. 7 and 8 of the Brooklyn Edison Company at Hudson Avenue), the main object was the measurement of the turbine and condenser performance by means of weigh tanks. The flow nozzle was a secondary consideration and was, therefore, neglected. It was not until the tests of the second unit that readings on the manometer were taken often enough. The points of the test on the second unit (No. 8) are in my opinion the only acceptable ones. In fact I would rather use only the last 6 of these. During these last 6 points the manometers were read every 1/2 min. If Mr. Cooper will consider only the points taken on the second unit (No. 8) the results will check our calibration curve much closer. These points are given in Table 2 of this discussion.

TABLE 2 FLOW COEFFICIENTS OF 12 IN. × 5 IN. NOZZLE AS DETERMINED BY WEIGH TANKS DURING A TURBINE TEST

$\log_{10}$ of Reynolds' number	Coefficient of discharge, C
6.125	0.991
6.145	0.995
5.931	0.999
6.056	0.998
6.041	0.986
6.261	0.997
6.286	1.002
6.441	1.001
6.405	1.001
6.124	1.000
6.145	0.997

TABLE 3 CALIBRATION RESULTS OF A 2.8812-IN. × 5.762-IN. FLOW NOZZLE (WATER TEMPERATURE, 69 F)

(Data by Prof. W. S. Pardoe)

Coefficient	Reynolds' number
0.9540	31310
0.9665	42490
0.9745	67090
0.9785	89450
0.9785	109580
0.9868	153190
0.9869	183380
0.9900	216920
0.9905	249230
0.9920	248230
0.9930	284010
0.9920	323150
0.9910	355570
0.9920	391350
0.9945	485280
0.9950	588150
0.9950	686540
0.9950	726800
0.9945	907940
0.9955	590300

In the use of flow nozzles for precise testing it is absolutely essential that the fluctuations in flow be slow enough for the manometer to follow the pressure changes and also for the observers to follow the manometer. I have not yet found a plant where these conditions could not be satisfied by some extra manipulation, as for example, either operating the pumps at different suction levels or using hand control of the flow.

It is true that, with respect to the location, the installation of the nozzle during the tests referred to by Mr. Cooper was entirely satisfactory but the conditions of flow during these tests were not. The flow fluctuated rather widely and rapidly.

Dr. Moss will be interested in the calibration results given in Table 3 on a new nozzle in which the pressure taps are brought straight out from the throat.

Mr. Dickinson's statement that the fluid nozzle is a practical device for testing turbines confirms my own experience.

Since writing this paper, I have obtained a calibration on a 2.8812-in. × 5.762-in. nozzle. This nozzle was welded into a 9-ft length of seamless steel tubing. It was made with four separate throat taps and a flat exit face very much like the nozzle shown in Fig. 12 of the paper. The nozzle and the tube were calibrated together. Table 3 gives the results of the calibration.

Flow Distribution in Forced-Circulation Once-Through Steam Generators[1]

H. J. Kerr.[2] The authors' paper confirms and extends the information presented by the writer in his paper,[3] "Once-Through Series Boiler for 1500 to 5000 Lb Pressure," in which the effect of inlet feedwater and outlet steam temperature on the instability of circuits was shown in diagrams. The value of resistances in stabilizing the flow and to compensate for unequal heat absorption in the different circuits was pointed out.

With reference to the authors' paper, the freedom from deposits in the test apparatus above 2500 lb pressure, irrespective of steam temperature, is worthy of note. Does this mean that above this pressure, steam to turbines will not need to show a purity represented by a resistance of 1,000,000 ohms to permit of continuous operation?

In determining the friction factor for a given Reynolds' number, the authors have used a straight-line projection on logarithmic coordinates of the known viscosity values of water up to 320 F. Probably this is a fair approximation though it does not agree, above 500 F, with Hevesy's values as given in Landolt and Bornstein tables. There may be some question as to the special point shown in Fig. 4 of the paper being discussed, checking in the case of steam as it apparently does with water.

Dealing with the question of stability in the boiler proper, the authors, in Fig. 8, show the effect of inlet-water temperature. These curves can be considered as a magnification of a small section of the curves in Fig. 7 of the writer's paper[3] previously referred to. I believe it would be clearer if the curves were extended over a greater temperature range, thus showing the reversal of direction which takes place, as it is, of course, impossible for the 200-F water curve to continue indefinitely in the direction shown, although it will continue in this direction until the tube is burned.

Fig. 9 of the paper under discussion shows the value of resistances in stabilizing flow. I have found, however, after talking to several engineers, that the significance of the dropping pressure with increasing enthalpy is not well understood. Per-

[1] Published as paper FSP-56-16 by H. L. Solberg, G. A. Hawkins, and A. A. Potter, in the November, 1934, issue of the A.S.M.E. Transactions.

[2] Babcock & Wilcox Company, New York, N. Y. Mem. A.S.M.E.

[3] "Once-Through Series Boiler for 1500 to 5000 Lb Pressure," by H. J. Kerr, Trans. A.S.M.E., vol. 54, 1932, paper RP-54-1a.